

Development of Noise Propensity Index (NPI) for Robust Brake Friction

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ABSTRACT

A semi-empirical index to evaluate the noise propensity of brake friction materials is introduced. The noise propensity index (NPI) is based on the ratio of surface and matrix stiffness of the friction material, fraction of high-pressure contact plateaus on the sliding surface, and standard deviation of the surface stiffness of the friction material that affect the amplitude and frequency of the stick-slip oscillation. The correlation between noise occurrence and NPI was examined using various brake linings for commercial vehicles. The results obtained from reduced-scale noise dynamometer and vehicle tests indicated that NPI is well correlated with noise propensity. The analysis of the stick-slip profiles also indicated that the surface property affects the amplitude of friction oscillation, while the mechanical property of the friction material influences the propagation of friction oscillation after the onset of vibration. An additional case study for noise reduction was carried out using commercial brake friction materials before and after the modification of NPI. The results of the case study based on in-vehicle noise tests indicate that a robust friction material can be designed by reducing the stiffness ratio and large contact plateaus on the sliding surface of brake friction materials.

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INTRODUCTION

Brake induced noise and vibrations have been considered as important issues since the early stage of vehicle development, and thus, efforts to reduce friction instability have been increased to meet the requirements of recent quiet vehicles. It is known that the friction oscillation during brake application has a wide range of frequencies. Therefore, multifaceted approaches are required to reduce brake noise and vibration due to frequency-specific triggering mechanisms. Much effort has been made to eliminate the brake-induced noise and vibrations from the judder, from low frequency to high frequency squeal. However, a comprehensive solution for the brake noise problems has never been successful, and the cure for brake noise has relied on empirical methods.

Various methodologies have been attempted to eliminate the squeal noise based on the fundamental understanding of squeal triggering and propagation. To explain the triggering mechanism of squeal, the stick-slip theory [1,2] and sprag-slip phenomenon [3] at the sliding interface have been investigated. In addition, various experiments based on modal analysis and numerical methods such as the complex

eigen value analysis to avoid the resonance of brake components have been performed to determine the conditions to avoid friction instability [4, 5, 6]. However, brake squeal still occurs frequently in the development stage of a vehicle and in the middle of mass production of a vehicle although the brake system is well tuned to prevent noise in the development stage.

The elusive nature of the brake squeal and difficulties in identifying the root cause of the problem are mainly attributed to the lack of fundamental understanding of the triggering mechanism of squeal and vibrations produced at the sliding interface. One of the well-accepted mechanisms of triggering friction-induced vibration is the self-excited friction oscillation known as stick-slip. The stick-slip, which is a series of friction cycles, is composed of a stick episode followed by a slip event in a cycle. The stick episode is related to adhesion at the interface, and the slip occurs when shear failure occurs at the interface due to the elastic energy accumulated during the stick episode [7,8]. The accumulated energy released during the slip episode can trigger noise and vibrations since the impact energy released at the slip event generates a hammering effect [9] on the brake system and can cause vibration of the components at their

natural frequency. In this case, the amplitude of the stick-slip can play an important role in triggering brake noise because a strong self-excited vibration can be triggered with the large amplitude of the stick-slip oscillation. Moreover, the topography of the friction material surface strongly affects the amplitude of the stick-slip oscillation [10]. A smooth friction material surface increases the stick-slip amplitude and aggravates the friction instability since the smooth surface is accompanied with high contact stiffness and high shear failure stress.

Brake noise can also be triggered by the resonance of the brake system due to friction oscillation since the frequency of the stick-slip decreases as a function of time during brake application. The velocity-dependent stick-slip frequency can also be explained by the change of the slope of the stick episode in the stick-slip profile [2]. Therefore, the stick-slip oscillation at a specific sliding velocity can intensify the self-excited vibration along with the components in the brake system. When self-excited friction oscillation initiates at the sliding interface, the oscillatory behavior is affected by the mechanical properties of the friction material surface. Analytical solutions for a simple sliding model indicated that tangential contact stiffness is important for the amplitude and frequency of stick-slip [11].

In this study, a semi-empirical index is designed based on the stick-slip phenomena to represent the noise propensity of brake friction material. Using mechanical properties such as contact stiffness from the surface and matrix of the friction material and practical contact area on the friction material surface, the noise propensity index (NPI) is introduced and the correlation between NPI and brake noise is evaluated. This study indicates that the properties of the friction material such as surface stiffness and size of the contact plateaus are crucial indicators for estimating the propensity of brake-induced noise and vibrations, suggesting that the NPI could be used as a preventive design standard for a robust brake system without noise and vibration.

DESIGN OF NOISE PROPENSITY INDEX (NPI)

The noise propensity index was designed based on the analytical model and experimental evidence. The semi-empirical index includes normal contact stiffness obtained from the friction material surface and matrix, which is based on the solution of the equation of motion in a sliding model [12]. The solution indicates that the ratio between the tangential contact stiffness from the sliding surface (k_{ts}) and the matrix (k_{tm}) affects the amplitude and frequency of the stick-slip oscillations produced during sliding. The mathematical solution indicates that the stiffness ratio (k_{ts} / k_{tm}) determines the stick-slip behavior: a pronounced stick-slip is observed for a large stiffness ratio. Stick-slip profiles obtained from the numerical solution of an analytical model, shown in Fig. 1, indicates that the oscillation amplitude decreased when the stiffness ratio was reduced. Experimental results [10] also reveal the strong effect of the stiffness ratio on the same friction materials with different surface roughnesses (Fig. 2).

The size of the contact area also affects the stick-slip amplitude since the friction coefficient of the local contact plateaus can increase due to the plastic deformation of the high-pressure surface. This is because the contact plateaus exposed to high pressures above the stress level for plastic deformation of the asperities increase the real area of the contact and the shear stress leading to interfacial failure at the slip events after stick episodes. The local large plateaus that emerge after burnishing can therefore play an important role in increasing the noise propensity of the friction material. Experimental evidences showing pronounced stick-slip phenomena in friction materials with large local plateaus confirm the reasoning behind the formulation of NPI. Fig. 3 shows the pronounced stick-slip oscillations when large contact plateaus are present on the sliding surface. Experimental results also showed the effect of the distribution of contact stiffness on the friction material surface. Since the friction material is not homogeneous, but composed of various ingredients, the stiffness distribution on the contact surface can be different and affects the amplitude and frequency of the stick-slip during sliding. The effect of stiffness distribution on noise occurrence is shown in Fig. 4.

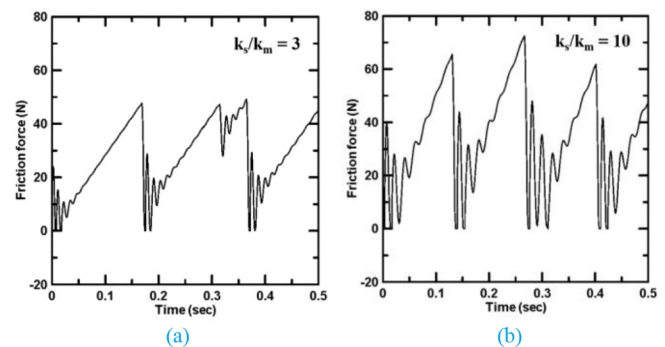


Figure 1. Stick-slip profiles obtained from the numerical solution of the analytical sliding model at a sliding velocity of 0.5 mm/s with different stiffness ratios: (a) $k_s/k_m=10$, (b) $k_s/k_m=3$.

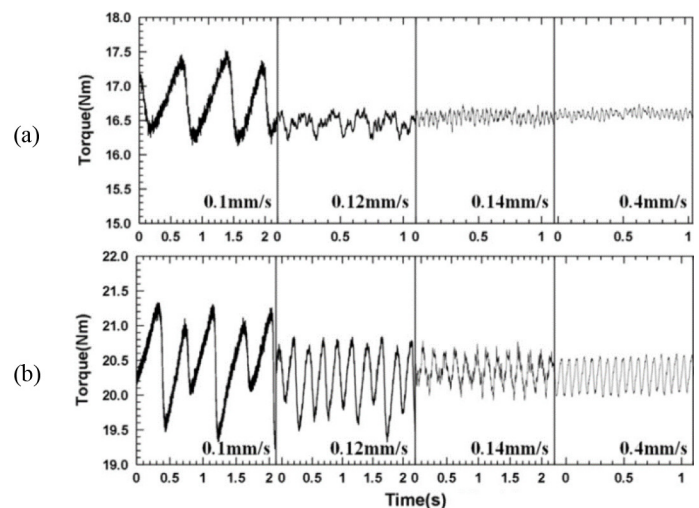


Figure 2. Stick-slip profiles obtained from the friction materials with different stiffness ratios: (a) $k_s/k_m = 9.4$, (b) $k_s/k_m = 31.8$

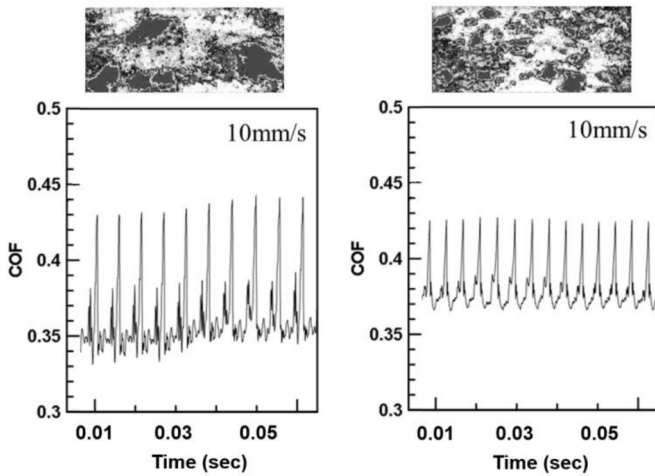


Figure 3. Stick-slip profiles obtained during low velocity drag test from friction material surfaces with (a) large and (b) small plateaus.

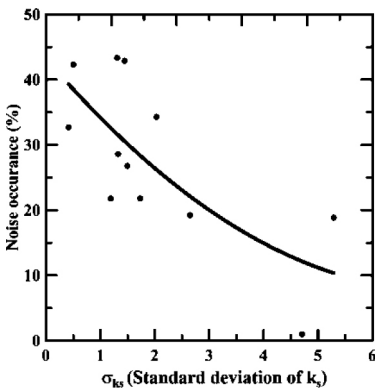


Figure 4. Noise occurrence plotted as a function of standard deviation of surface stiffness k_s .

Based on the four material properties, a semi-empirical index for noise propensity was designed to estimate noise occurrence during brake applications. Assuming that the stiffness ratio (k_{ts}/k_{tm}) and fraction of the contact plateaus increase the amplitude of stick-slip and the standard deviation of the surface stiffness decreases the stick-slip amplitude, the noise propensity index was formulated as:

$$NPI = \left(\frac{k_s}{k_m} \right) \cdot \left(\frac{R_p}{\sigma_{ks}} \right), \quad (1)$$

where k_s and k_m are the contact stiffnesses of the friction material surface and matrix, respectively, R_p is the fraction of local plateau areas above the average pressure, and σ_{ks} is the standard deviation of surface contact stiffness on the friction material surface.

EXPERIMENTAL METHODS

Contact Stiffness

Contact stiffness of the friction material was measured by using a sapphire hemisphere (diameter = 5 mm) to minimize the possible misalignment of a top platen used to obtain the load-displacement plot. Although the stick-slip was affected by tangential stiffness,

normal stiffness was measured for convenience since the tangential stiffness can be obtained from the normal stiffness using the Poisson ratio of the material [13]. From the load-displacement plot, stiffness as a function of pressure was obtained, which was composed of surface and matrix stiffness of the friction material. Fig. 5 shows the schematic of the stiffness measurement and the resultant contact stiffness as a function of pressure.

The stiffness behavior observed in Fig. 5 (c) is attributed to the contributions from the surface and matrix. Assuming that the other mechanical contributions from the measuring device are same, the surface stiffness (k_s) and matrix stiffness (k_m) can be calculated from the normal contact stiffness using the following relation since they are connected in series [14].

$$k_n = \frac{k_s \cdot k_m}{k_s + k_m}, k_s = A \cdot P, k_m = B \cdot P + C \quad (2)$$

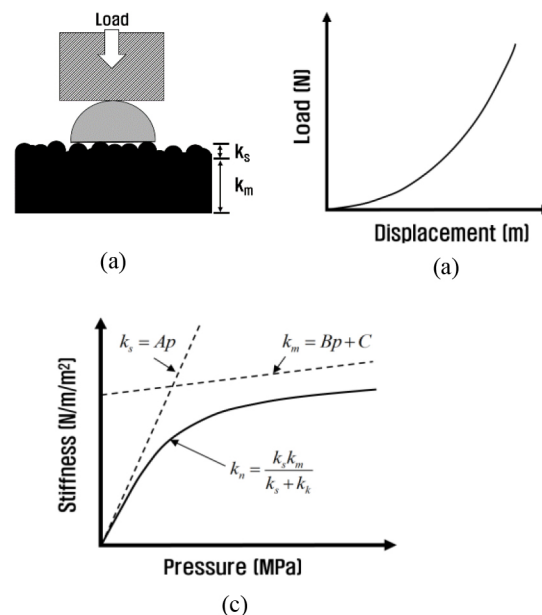


Figure 5. (a) Schematic of the indentation test using a hemisphere and load-displacement plot. The stiffness-pressure plot (c) calculated from the load-displacement plot (b).

Distribution of Contact Stiffness

The distribution of surface contact stiffness was examined by measuring k_s at different locations on the friction material surface, which was presented as the standard deviation. Eighteen measurements were performed for each specimen without overlapping the contact area to obtain the standard deviation of stiffness variation. An example of stiffness variation is shown in Fig. 6. The distribution of surface contact stiffness was obtained from the variation of the load-displacement curves using the relation between k_n , k_m , and k_s from Eq. 2.